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COMPRESSOR BLADE MONITORING SYSTEM FOR A VA1310 (ALLIS CHALMERS) WIND TUNNEL COMPRESSOR

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TECHNICAL REPORT
FINAL REPORT FOR PERIOD JULY 1979 THROUGH MAY 1980

FLIGHT DYNAMICS LABORATORY
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The purpose of the work summarized in this report is to identify and develop a cost effective, reliable procedure for determining potential blade failure in time to prevent actual occurrence. The procedure is developed for application to an Allis-Chalmers ten stage, axial flow compressor Model VA1310. The approach followed in conducting this study included a review of the current techniques used to insure blade integrity, a review of other approaches as described in litterature for verifying the condition of the compressor blades and, finally, development of a technique suitable for use with the VA1310 compressor.

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FORWORD

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INTRODUCTION

The measurement of vibration levels of rotating machinery is the general approach used for verifying machinery health and signs of impending failure. In axial flow compressors, however, a common cause of failure, which cannot be detected until after the fact, is cracking and ultimate breaking of compressor blades. If failure occurs in the early compression stages (a common occurrence, since these are the longest blades) considerable damage can occur as blade fragments pass through the latter stages. A need exists, therefore, to detect impending failure of blading before serious damage is encountered.

The purpose of the work summarized in this report is to identify and develop a cost effective, reliable procedure for identifying potential blade failures in time to prevent the actual occurrence. The procedure is developed for application to an Allis-Chalmers ten-stage, axial flow compressor, Model VA 1310. The approach followed in conducting this study included a review of the current techniques used to insure blade integrity, a review of other approaches as described in literature for verifying the condition of compressor blades and, finally, development of a technique suitable for use with the VA 1310 compressor.

SECTION II

DISCUSSION

The compression of air or gas through an axial flow compressor is accomplished with flow passing axially through a number of blading stages -- each stage containing blades of shorter length to decrease the volume of the gas as it proceeds through the compressor. Between each row of rotating blades, a stationary blade row is used to redirect the gas to the next rotating blade stage. The rotating blades are subjected to three types of forces during operation; i.e., rotating centrifugal force, gas forces and thermal forces resulting from the energy losses during compression of the gas. Blade design takes into account all of these steady-state forces to ensure adequate strength and reliability. It is the dynamic gas forces, however, that often cause ultimate damage of the blades. These forces can occur from unstable gas flow such as that encountered at high discharge pressure where surging, characterized by sharp pulsations of flow, can induce blade vibration and ultimate fatigue.

Periodic pressure pulsing that coincides with the blade natural frequency can also induce vibration and fatigue of the blade. Although compressors are designed to avoid operating conditions that would induce blade vibration, occasional off-design conditions may be encountered that could lead to blade fatigue. Blade damage can also be induced from foreign objects passing through the compressor. If blade cracking or damage does occur, there is presently no acceptable technique to detect the damage other than disassembly and inspection of the blades. It is the purpose of this study to review the techniques that have been applied for detecting blade damage, and develop an approach that could be applied to an Allis Chalmers VA 1310 axial flow centrifugal air compressor that eliminates the need for disassembly.

1. VA 1310 Compressor Description

Since the blade inspection or monitoring system technique is being developed for a specific compressor, the following section provides a brief description of the compressor blading configuration. The VA 1310

axial flow compressor contains a drum type rotor (constant blade root diameter) and decreasing external diameter of the installed blade stages. The blade root is a conical section that is bolted to the drum. Figure 1 depicts a typical compressor blade installation illustrating the blade root connection. The compressor contains ten similar stages each having a different blade height. The first four stages of blades utilize the same blade configuration with varying length and the remaining six stages use the same blade design except for chord length and blade length. The stators between each rotating stage are variable in pitch, externally controlled. The following characteristics describe the operating parameters for the compressor:

Allis Chalmers 10-stage Axial	Compressor
Design Operation	(3600 RPM)
154000 s.c.f.m at 70°F;	14.7 psia
Delivery Pressure	30.0 psia
Maximum Temperature	465°F
Installed Horsepower	8500
Momentary Maximum Horsepower	11000
Inlet Temperature	100°F
Blading	38 ins. root diameter
Rows 1-4	37 Blades
Rows 5-10	47 Blades

2. Existing Blade Inspection Techniques

The approach presently used to check the condition of the compressor blades requires complete disassembly of the compressor and removal of the rotor. Two techniques are applied to verify blade condition. The first approach requires instrumentation of each blade with a miniature accelerometer, and impacting the blade to cause it to resonate at its natural frequency. The frequency of the first cantilever mode is determined in this manner. If a crack exists at the base of the blade, the natural frequency will differ from the frequency of a new blade.

The actual procedure followed in ringing the blades was to utilize an instrumented hammer containing an accelerometer. This signal was used to trigger an FFT analyzer to capture the impact and blade accelerometer output. A frequency spectra of transfer function (ratio of blade accelerometer output to hammer accelerometer input) was used to determine the blade natural frequency.

Blade frequencies were recorded in 1973 and compared to blade frequencies recorded in 1978. The difference in frequency between these two time periods for the first blade row ranged from -.9 Hz to + 1.7 Hz, about a frequency of 359 Hz. If a crack developed, the blade frequency would decrease and therefore the + 1.7 Hz frequency is an indication of the repeatability of the method. This trend of frequency variations was not consistent for all blade rows. In the case of the tenth-stage row, for instance, the resonant frequency dropped on all blades between the 1973 and 1978 time periods. The decrease in frequency ranged from 4 Hz to 8.1 Hz, about an average frequency of 1033 Hz. Since the magnetic particle inspection indicated no blade cracks, these variations in frequency cannot be attributed to blade damage. Variations in measured frequency may be associated with differences in measurement techniques or changes in the blade anchoring conditions. Based on these results, it would appear that the accuracy of this technique of checking for blade resonant frequency is ±0.5% of the resonant frequency. Estimated shifts in resonant frequency prior to failure (see Section 4) range from 0.7% to above 1.5% depending upon crack location, size and stage. Measurement of blade-resonant frequencies therefore may not be sufficient to ensure the integrity of the blades in the VA 1310 compressor.

Ringing of the compressor blades and metal particle inspection necessitate disassembly of the compressor. Estimates for this work included four men for one week of disassembly, one week for inspection, and one and one-half weeks for reassembly. Neglecting materials and downtime, it is estimated that the cost for this effort exceeds $16 K. This procedure is repeated after approximately every 2000 hours of operation.

A monitoring system to avoid these costs, increase availability, and provide an earlier indication of blade failure, should include the following:

- Capability to monitor blades during machine operation.
- High reliability for long life.
- Minimal complexibility.

These were the goals that were applied in developing a recommended approach for an improved blade inspection or monitoring technique.

3. Compressor Blade Analysis

The most common cause of blade failure in axial flow compressors is the development of cracks in the high stressed areas of the blade which eventually leads to failure. Such cracks are generally introduced from aerodynamic excitations that force the blade to vibrate at one or more of its natural frequencies resulting in fatigue. The highest stressed region of the blade occurs at the blade root. If cracking of the blade does occur, blade deflections due to centrifugal and aerodynamic forces increase. The ability to measure blade deflections during operation could provide a means of monitoring for blade damage. In order to evaluate the feasibility of measuring blade deflections, an estimate was made of the order of magnitude of such deflections. A rigorous three-dimensional stress analysis was not considered necessary since complete details of blade curvature were not available. Available information of blade height width, thickness, chord height and blade natural frequency in the first cantilever mode was considered adequate to provide reasonable estimates of blade deflections. Unlike many axial compressors, the blade attachment method used in the VA 1310 compressor as shown in Figure 1 results in anchoring the blade over a portion of the blade root in lieu of the full width of the blade. The cross sectional inertia of the root for blade bending or torsion therefore cannot be calculated as a full width blade or from the area of the anchor point. In order to estimate the inertia

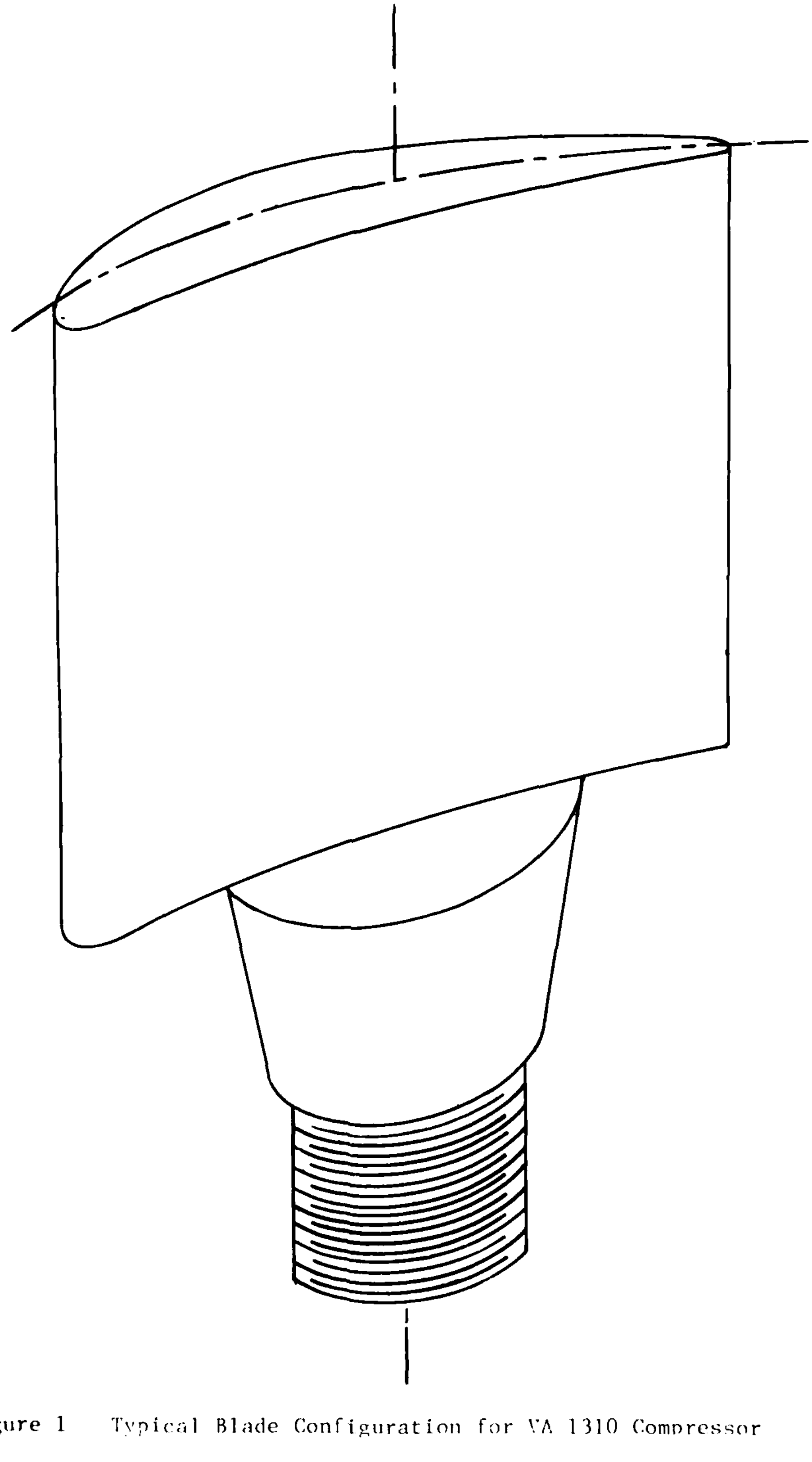


Figure 1 Typical Blade Configuration for VA 1310 Compressor

(which is required for any blade deflection analysis), the measured natural frequencies were utilized.

For a uniform cantilever anchored at the base the first bending mode may be defined as:

$$f = 0.56\sqrt{\frac{EI}{m\ell^3}} \tag{1}$$

where

E = mod. of elas. (Psi)
I = inertia in^4
m = mass (no. of sec^2/in)
ℓ = length
f = natural freq. (Hz)

Solving for inertia (I):

$$I = 2.79f^2 w\ell^3 x10^{-10}$$

where

w = weight (lbs)

Blades in rows 1 through 4 are the same blade of four-inch width with different lengths. Blade thickness and chord height of both blade types were furnished by Allis Chalmers to permit calculation of blade weight. With knowledge of blade weight, length, and average natural frequency, the inertia of the blade was calculated with the following results:

Blade No.	I (in^4) (in bending)	
1	.0363	
2	.0363	
3	.0378	Avg I = .0368
4	.0368	
5	.0085	
6	.0085	
7	.0081	
8	.0083	Avg I = .0082
9	.0080	
10	.0080	

Figure 2 depicts a blade profile and provides an expression for the bending inertia of the blade root from Reference [1]. This equation estimates the inertia for the entire blade root. For blades 1 through 4:

b - 4.0 inch

t - 0.573 inch (max. blade thickness)

h - 0.375 inch (chord height)

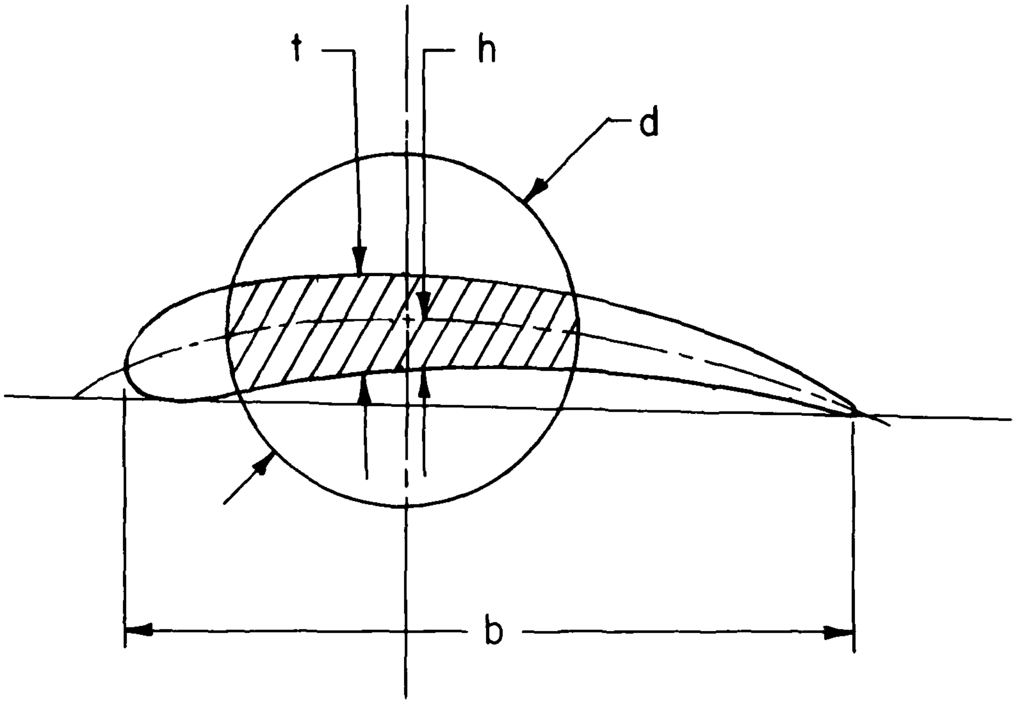
$I = 0.041 \cdot 4 \cdot .573(.573^2 + .375^2)$

$I = 0.04407 \text{ in}^4$

The inertia of the only crosshatched area may be estimated from

$$I = \frac{bt^4_{avg}}{12} = \frac{2 \times 0.53^3}{12} = .0248 \text{ in}^4$$

As anticipated, the inertia of the blade found from ringing the blade (.0368 in^4) falls between the extremes of that computed for a complete blade root cross section and that computed for just the root anchor point. The inertia, however, is closer to that of a complete blade. In either case, if a crack occurs at the root, the reduction in inertia is a first-order effect on reduction in blade width (b). In one case the width of



$$I = .041\,bt(t^2 + h^2) \qquad \text{Ref (1)}$$

Figure 2 Plan View of Blade

the blade is overestimated and in the other case, it is underestimated. For analysis purposes in bending, the inertia values of .0368 in^4 and .0082 in^4 were utilized.

a. Blade Deflection in Bending

In order to determine the magnitude of bending of the blade, an estimate of the blade loading is required. For this estimate it is assumed that equal work is performed in each stage of compression. If it is assumed that the nominal air inlet temperature is $100^{o}F$ at 14.7 PSIA, and an air flow of 154,000 SCFM at a discharge pressure of 30 PSIA occurs, the adiabatic discharge temperature is:

$$\frac{T_o}{T_i} = \left(\frac{P_o}{P}\right)^{\frac{K-1}{k}} \qquad (2)$$

where

T = absolute temperature (^{o}R)

P = absolute pressure (PSIA)

K = ratio of specific heats = 1.406

$$T_o = 560 \left(\frac{30}{14.7}\right)^{.283} = 685^{o}R$$

$$\Delta T = 125.3^{o}F$$

Assuming an adiabatic efficiency of 82.5% including inlet and discharge losses, the temperature rise is $152^{o}F$.

The aerodynamic horsepower is:

$$HP = 1.414\ WC_p \Delta T \qquad (3)$$

w = flow (lb/sec) = 191.6

C_p = specific heat = 0.241

$$HP = 1.414(191.6)\ .241(134.7) = 9914$$

Since this is greater than the installed horsepower, the conservative approach is to consider that 92.5% of the installed horsepower is representative of the aerodynamic horsepower during normal operation at 3600 RPM. The assumed horsepower, therefore, is 7863 or 786.3 HP per stage of compression. At 3600 RPM, the torque per stage is

$$T = \frac{63000\ HP}{N} = 13759.4 \text{ in. lbs.} \qquad (4)$$

For the first compressor stage, the blade length is 7.305 in. and there are 37 blades resulting in a blade force of:

$$F = \frac{T}{R \times n}$$

$$R = \frac{7.305 + 38}{2} = 22.6525 \text{ in.}$$

$$n = 37$$

$$F_1 = 16.42 \text{ lbs. per blade}$$

For the tenth compressor stage, blade length is 3.293 in. and there are 47 blades, resulting in a blade force of:

$$F_{10} = \frac{13759.4}{\frac{(3.293 + 38)}{2} 47} = 14.18 \text{ lbs.}$$

Both these computed forces are tangential forces, the normal bending forces being a function of the blade angle of attack; i.e.,

$$F_1 = \frac{16.42}{\cos 39^\circ 16'} = 21.21 \text{ lbs}$$

$$F_{10} = \frac{14.18}{\cos 34^\circ 22'} = 17.18 \text{ lbs.}$$

Considering the blade as a cantilever, the force is distributed over the cantilever length uniformly. For a cantilever beam with uniformly distributed load, the deflection at the tip is:

$$y = \frac{w\ell^3}{8EI} \tag{5}$$

$$y_1 = \frac{21.21\ (7.305)^3}{(8)\ 29.6 \times 10^6\ (.0368)} = .95 \times 10^{-3} \text{ in.}$$

$$y_{10} = \frac{17.18\ (3.293)^3}{(8)\ 29.6 \times 10^6\ (.0082)} = .32 \times 10^{-3} \text{ in.}$$

During operation, centrifugal forces tend to reduce the magnitude of blade bending and, therefore, actual amplitudes would be smaller than computed. The aerodynamic bending forces are steady state forces but are indicative of the possible magnitude of dynamic forces that could be encountered at blade resonance if blade stall occurred.

The monitoring of static blade deflection due to aerodynamic moments does not appear to offer promise as a monitoring technique. Oscillation of the aerodynamic forces during blade stall or compressor surge may be adequate to excite the blade into resonance permitting amplification of these forces and thus allowing monitoring of the dynamic motions of the blade. Further discussion regarding measurement of blade dynamics is given in Sections 4 and 5.

b. Blade Deflection in Torsion

Since the center of pressure on the blade face does not coincide with the center of gravity, an aerodynamic moment is introduced into the blade. This moment acts to produce a torsional deflection of the blade. Assuming a linear pressure rise across the blade, the moment arm between the center of pressure and center of gravity would be 1.066 inches for the first-stage blade and 0.866 for the tenth. The resultant moment on the blade is 22.61 in. lb. for a blade in the first stage and 14.88 in. lb. for the tenth-stage blade. The angular blade twist is defined as:

$$\theta = \frac{ML}{IG} \tag{6}$$

where

M = applied moment (in. lb.)
L = one-half blade height (in.)
$I = 0.281\ b\ \delta^3\ (\mathrm{in}^4)$
b = blade chord length (in.)
δ = blade thickness

$$\theta_1 = \frac{22.61 \times 3.6525}{.061 \times 12 \times 10^6} =$$

$$\theta_1 = .114 \times 10^{-3} \text{ radians}$$

Tip tangential deflection $y_1 = .23 \times 10^{-3}$ in.

$$\theta_{10} = \frac{14.88 \times 1.6465}{.0177 \times 12 \times 10^6}$$

$$\theta_{10} = .189 \times 10^{-3} \text{ radians}$$

Tip tangential deflection $y_{10} = .369 \times 10^{-3}$

As in the case of static bending deflections of the blade due to aerodynamic loading, aerodynamic induced twist deflections are also small, and do not offer promise as a monitoring technique unless amplified by resonance of the blade.

c. Deflections of Blades Due to Centrifugal Loading

Figure 3 illustrates a blade containing a crack at the blade root. The crosshatched region indicates the area that is not cracked. The crack results in a shift between center of gravity of the blade and the reaction center at the root. When the blade root is cracked as shown in this figure, a bending moment is introduced to tilt the blade from the trailing edge toward the leading edge. Because of the blade rigidity in this direction, deflections due to this moment are small and have been ignored for analysis purposes. A second moment results due to the fact that the blade is not installed parallel to the rotating axis, but at a "setting angle" relative to the rotating axis. This produces a twisting moment about the "y" axis.

The twisting moment has a force of F sinθ at a moment arm of C cosθ from Figure 3; i.e.,

$$M = FC \sin\theta \cos\theta$$

where

F	= centrifugal blade force: lbs.
C	= 0.5 crack length: ins.
θ	= setting angle of blade: degrees
subscripts	= blade stage

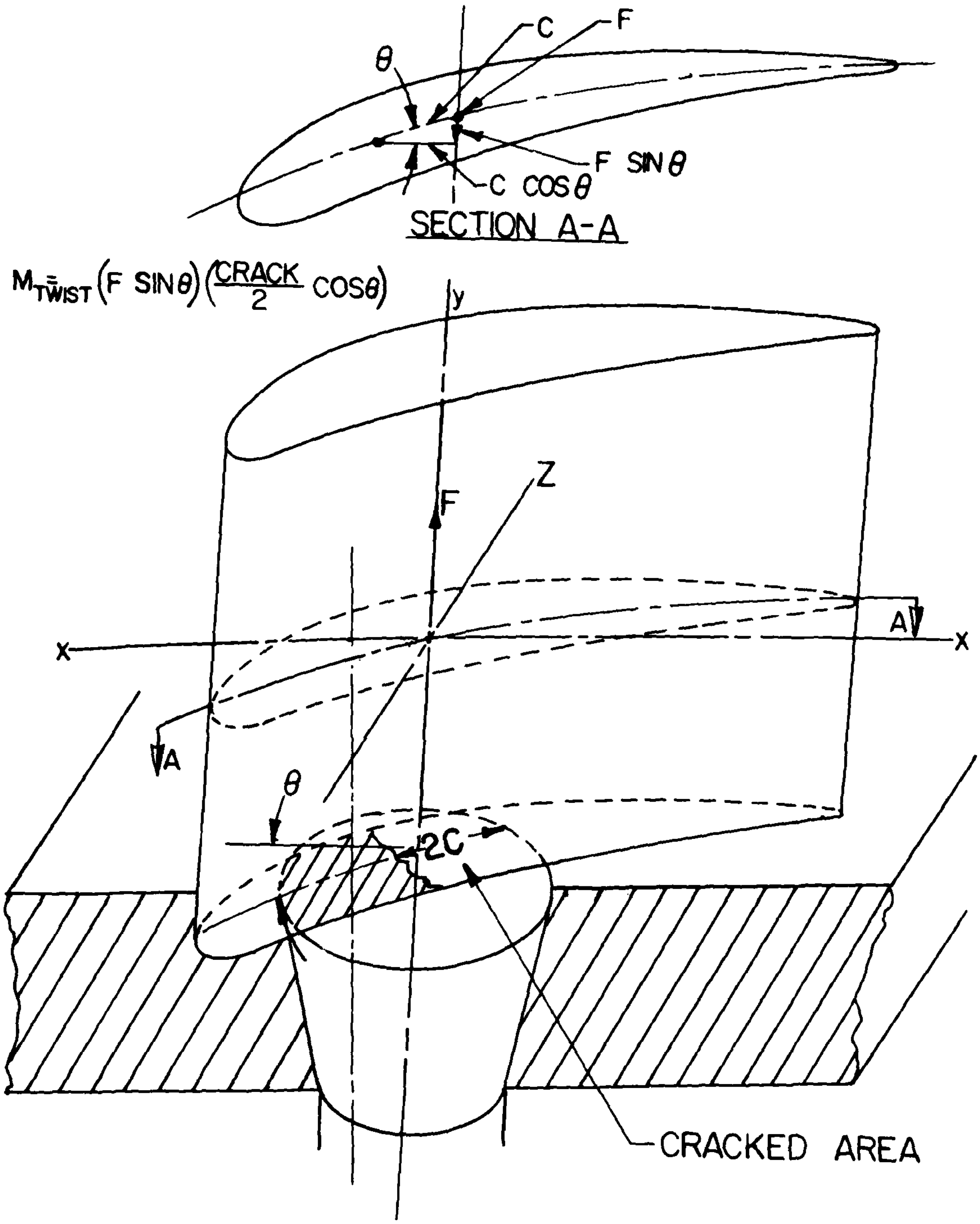


Figure 3 Torsional Moment Due to Blade Root Cracking Along Width of Blade

The centrifugal load at the blade root for the first-stage blade at 3600 rpm is 21,813 lbs. and for the tenth-stage blade is 5714 lbs. With a crack length C of 0.1 inch, the twisting moments are

$$M_1 = 21{,}813\ (.05) \sin 39^{\circ}16' \cos 39^{\circ}16'$$
$$M_1 = 534.4 \text{ in lbs.}$$
$$M_{10} = 5714\ (.05) \sin 34^{\circ}22' \cos 34^{\circ}22'$$
$$M_{10} = 133.1 \text{ in. lbs.}$$

These moments compare with the aerodynamic moments of 22.61 in. lbs. and 14.88 in lbs. for blade rows 1 and 10, respectively. The moments produce measurable deflections (5.44 mils and 3.3 mils) as compared to the deflections introduced aerodynamically. It should be noted that development of a crack in the root of the blade other than shown in Figure 3 results in a twisting moment on the blade that is computed in the same manner. The orientation of the bending moment on the blade, however, will change as crack location changes. In the case of a crack in the direction of Figure 4, the bending moment FC will introduce the maximum blade bending deflection. It is seen that blade deflections in the bending mode and twisting mode are induced from centrifugal loading when a crack in the root region occurs. The magnitude of the bending deflections will depend upon crack orientation.

If it is assumed that the maximum resolution of a measuring system, detecting blade deflection, is 0.0005 inches, an estimate may be made of the detectable crack size. For the first-stage blade, a crack of 0.0094 ins. could be discerned by measuring blade deflection due to twist and 0.030 in. crack for blade 10. If a crack occurred along the length of the blade root, the resulting bending moment would induce a deflection of the blade. The approximate crack size required to detect 0.0005 inch deflection is 0.0045 ins. for the first-stage blades and 0.0189 ins. for the

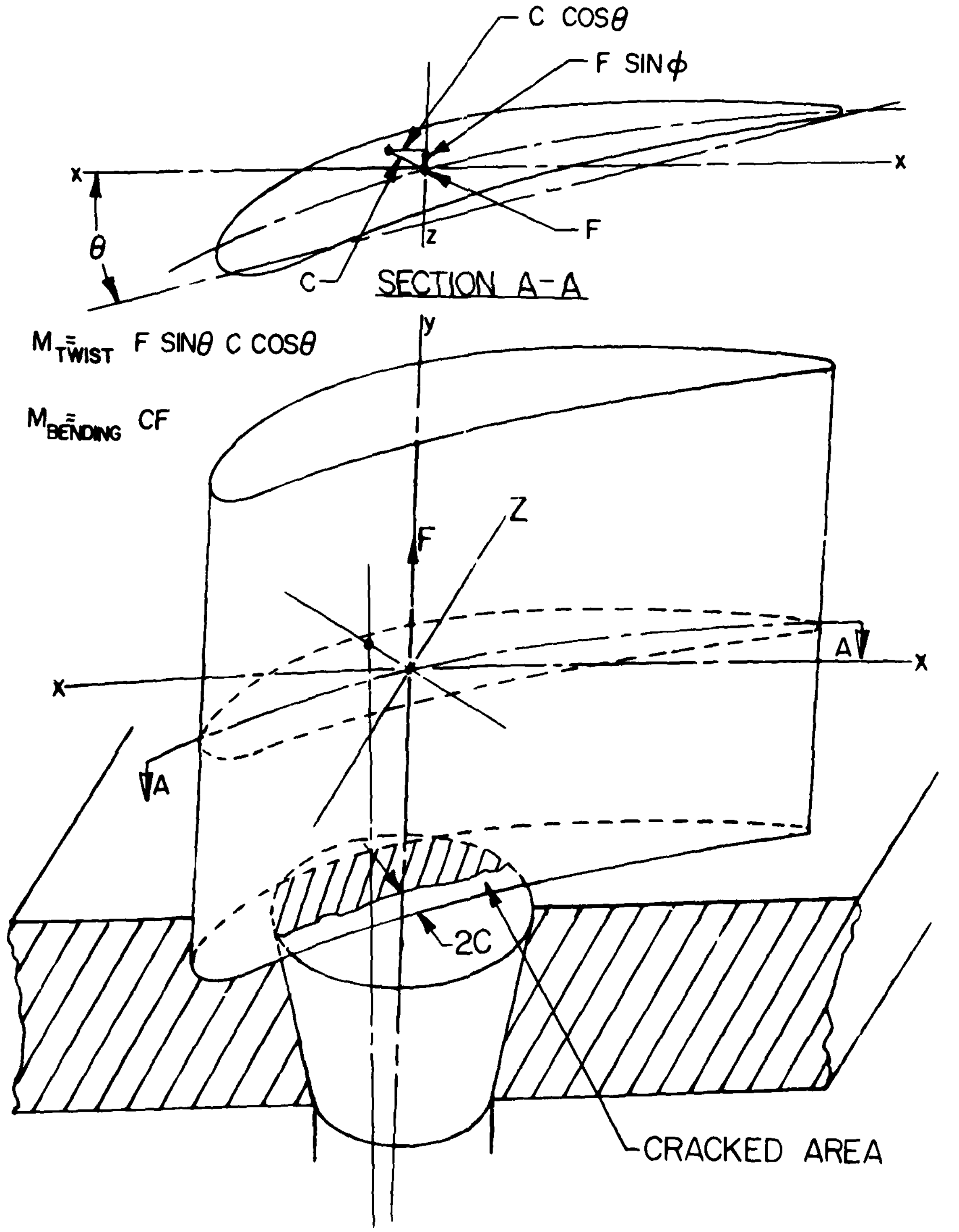


Figure 4 Moments Induced in Blade Due to Blade Root Cracking through Blade Thickness Direction

tenth-stage blades. In actual operation, the crack at a blade root would probably not progress along a single plane and blade deflection would be a combination of torsion and bending.

d. Summary of Blade Deflections

Aerodynamic loading on the axial flow compressor blades introduces both bending and torsion of the blades. The larger deflections are induced in the bending mode. Cracking in the blade root region will not significantly influence these deflections induced aerodynamically, until the crack size becomes sizable. This is due to the fact that changes in cross-sectional properties in the cracked region cannot be computed directly with changes in crack size. These properties do not make a step change between the cracked area and uncracked area of the blade. Some assumptions must be made regarding the transition of stress and deflection in this region. The conservative approach selected was to assume that blade properties do not change significantly to influence tip deflections until yielding occurs. The same assumption is made in estimating blade tip deflections induced by centrifugal loads after a crack at the blade root occurs. This permits comparison of the magnitude of deflections induced aerodynamically versus centrifugally.

If a crack does occur in the blade root, centrifugal forces induce moments in the blade that are measurable in terms of deflection at the blade tip in both bending and torsion neglecting changes in cross-sectional inertia properties. Such deflections can be discerned well before yielding of the blade in the root region is encountered. Centrifugal tensile stress in the root region range from 23,500 PSI in the first-stage blades down to 12,100 PSI in the tenth stage. Although the exact heat treatment of the 403 stainless steel blade material is not known, it is assumed that the yield strength of the material is in the range of 60,000 to 70,000 PSI. For a crack at the blade root of 0.100 in. in the

thickness direction of a first-stage blade, the estimated combined bending, tension and shear stress are estimated at 61,000 PSI. For a tenth-stage blade the estimated stress under the same conditions is 43,000 PSI. If displacement measurements of the blade tip during operation with a resolution of 0.5×10^{-3} ins. is possible, then detection of cracks from 0.010 to 0.100 in size can be made on the first-stage blade row and 0.030 to greater than 0.100 on the tenth-stage blade row before failure is anticipated.

4. Blade Vibrations

If blades are excited by an impact or shock wave, they will vibrate at their natural frequencies. The amplitude of vibration will decay with time depending upon the available damping in both the blade and the available aerodynamic induced damping. When positive damping is not available. sustained vibration of increasing amplitude will occur. This type of instability is often referred to as flutter. Other sources of blade excitation include turbulence in the air flow and blade natural frequencies coincident with nozzle passing frequency or other discrete forcing frequencies. Cracking in the root of a blade would result in a change in stiffness properties of the blade and reduce its natural frequency if excited. This change in frequency was one of two static inspection techniques used during blade inspection as previously discussed. It is conceivable that if a means of exciting the blade was available and the frequency could be measured, that the technique of measuring blade natural frequency could be accomplished during operation of the compressor.

In order to consider the use of blade vibrational frequency measurements during operation, it is desirable to provide an estimate of the anticipated shift in frequency with crack size to determine the type of resolution that would be required in a measuring system. In attempting to analyze the influence of a crack, some assumptions must be made regarding the inertia properties of the blade in the vicinity of the crack. For the analysis, a cylindrical cantilever beam was analyzed with different crack depths at the root of the cantilever. As shown in Figure 5, it was

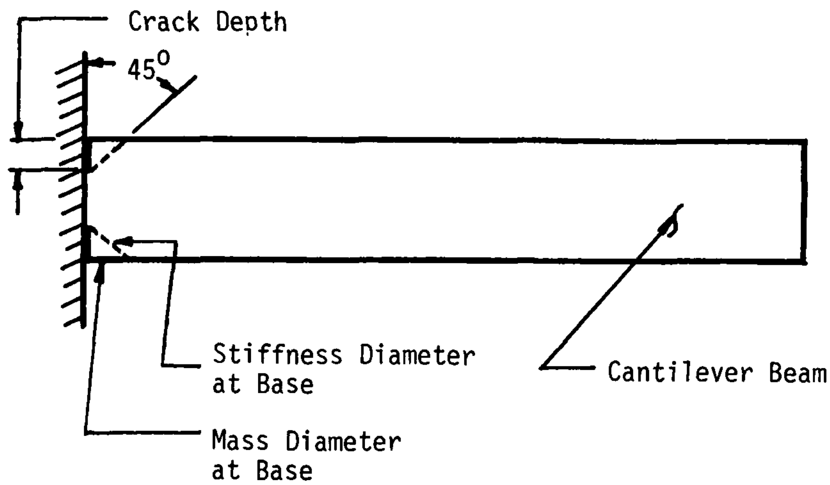


Figure 5 Model for Determining Influence of Crack on Beam Natural Frequency

assumed that cross-sectional properties of the beam followed a forty-five degree angle from the base of the crack outward. The natural frequency was computed using this model for various crack depths. The results of ratio of uncracked-to-cracked natural frequency were plotted as a function of the ratio of the cracked-to-uncracked cross sectional inertia properties in bending in the plane of the crack. These results are shown in Figure 6.

For a blade in the first-stage row, the cross-sectional inertia was estimated as 0.0368 in^4 from static resonance tests. The inertia is defined analytically as.

$$I = 0.041\ bt(t^2 + h^2) \qquad \text{Ref. [1]}$$

where

b = equivalent blade chord length (in.)
t = maximum blade thickness (in.)
h = chord height (in.)

For a crack reducing the blade thickness (t) by 0.100 inch, the blade stress will approach the yield limit of the material. This size crack results in a ratio of cross-sectional inertia properties of

$$\frac{I\ \text{cracked}}{I\ \text{uncracked}} = .641$$

From Figure 2.6, the estimated reduction in the first cantilever mode natural frequency is estimated as:

$$359\ \text{Hz} - (.993 \times 359\ \text{Hz}) = 2.513\ \text{Hz}$$

For the tenth-stage blade, a frequency decrease of 13.5 Hz would be anticipated for a 0.1 inch crack through the root thickness. Although this analysis provides only an estimate of the influence of cracks on

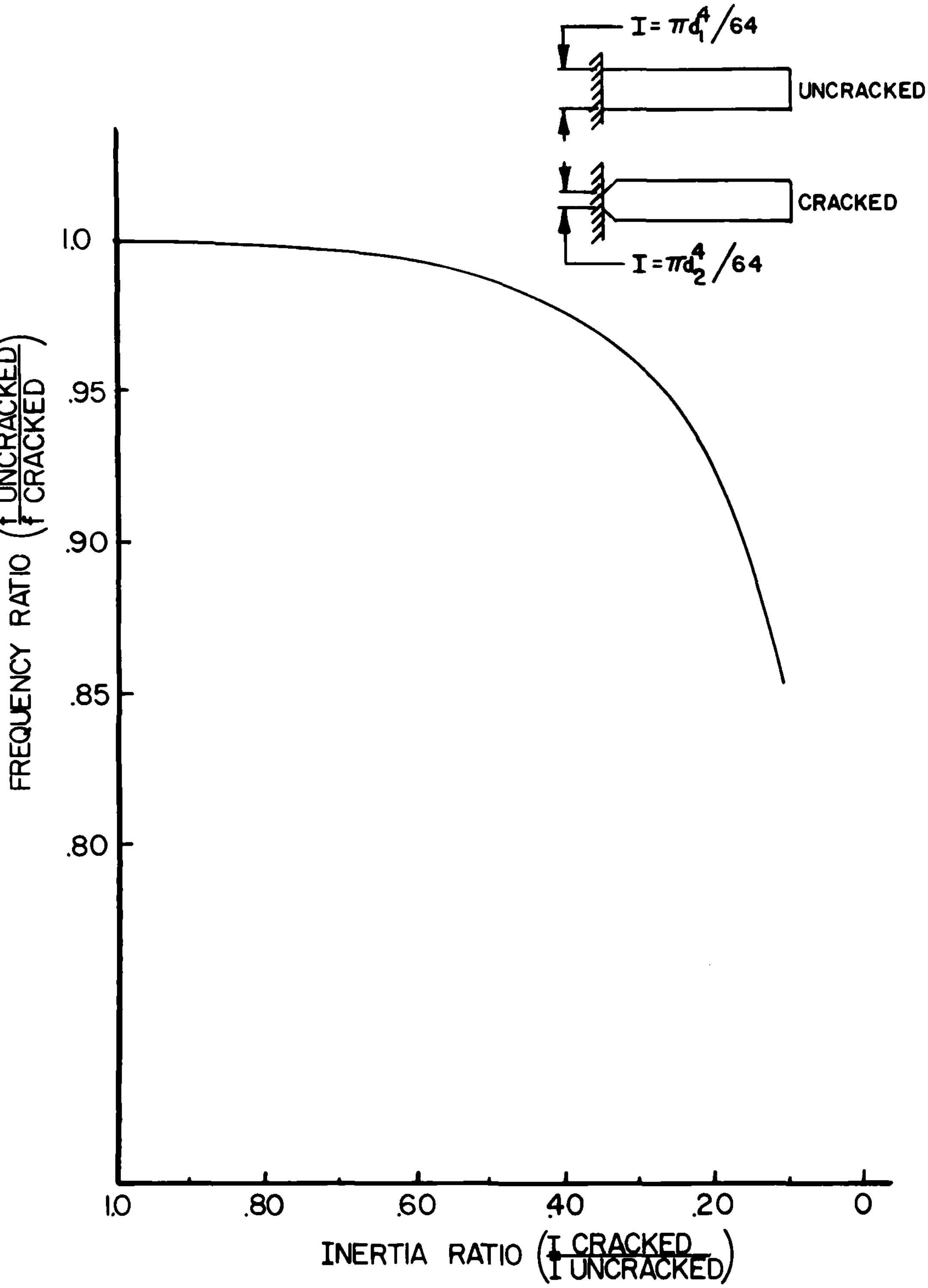


Figure 6 Influence of Crack Size on Natural Frequency of Cantilever

resonant frequency, it does indicate that frequency shifts due to cracks developed in the blade root region will not be large. A continuous monitoring system for the VA 1310 compressor, based on a shift in resonant frequency of the first cantilever mode would require frequency resolution in the range of 1 Hz. The shift in resonant frequency will be a function of crack location and the resonant mode that is being monitored. Higher bending modes or torsional modes might be more suitable for detecting blade cracks. The higher blade frequencies, however, pose more difficult instrumentation problems when considering monitoring of the blades during operation of the compressor.

In addition to the question of feasibility of using a frequency shift of the first cantilever mode to protect the blade from failure, there is also the question of exciting a blade into resonance during operation. The blades can be excited either mechanically (e.g., a trip-hammer) or aerodynamically by deliberate stalling of the blade. When a blade stalls, largely the whole of its aerodynamic lift is removed reducing the aerodynamic bending force to zero. In the case of the first-stage blade, this amounts to an approximately 22-pound-force that can induce resonance if the time transient is short. When individual stages stall, the stall is a rotating type stall where individual blades alternately stall and compress air. This implies the blade vibration will be inconsistent and difficult to measure. Alternately, it is possible to stall all stages simultaneously which is generally termed surging the compressor. This results in a thrust reversal on the machine and could damage the thrust system. Therefore, surging is not a recommended method of excitation.

The two problems of having sufficient frequency shift of blade resonance frequency prior to failure and developing a method of excitation of the blade, makes measurement of blade frequencies a questionable monitoring approach. Most systems that are directed toward measurements of blade vibrations are used to develop design criteria to avoid operating at conditions that would induce vibration. They are not concerned with small shifts in blade resonant frequency but frequency ranges to be avoided.

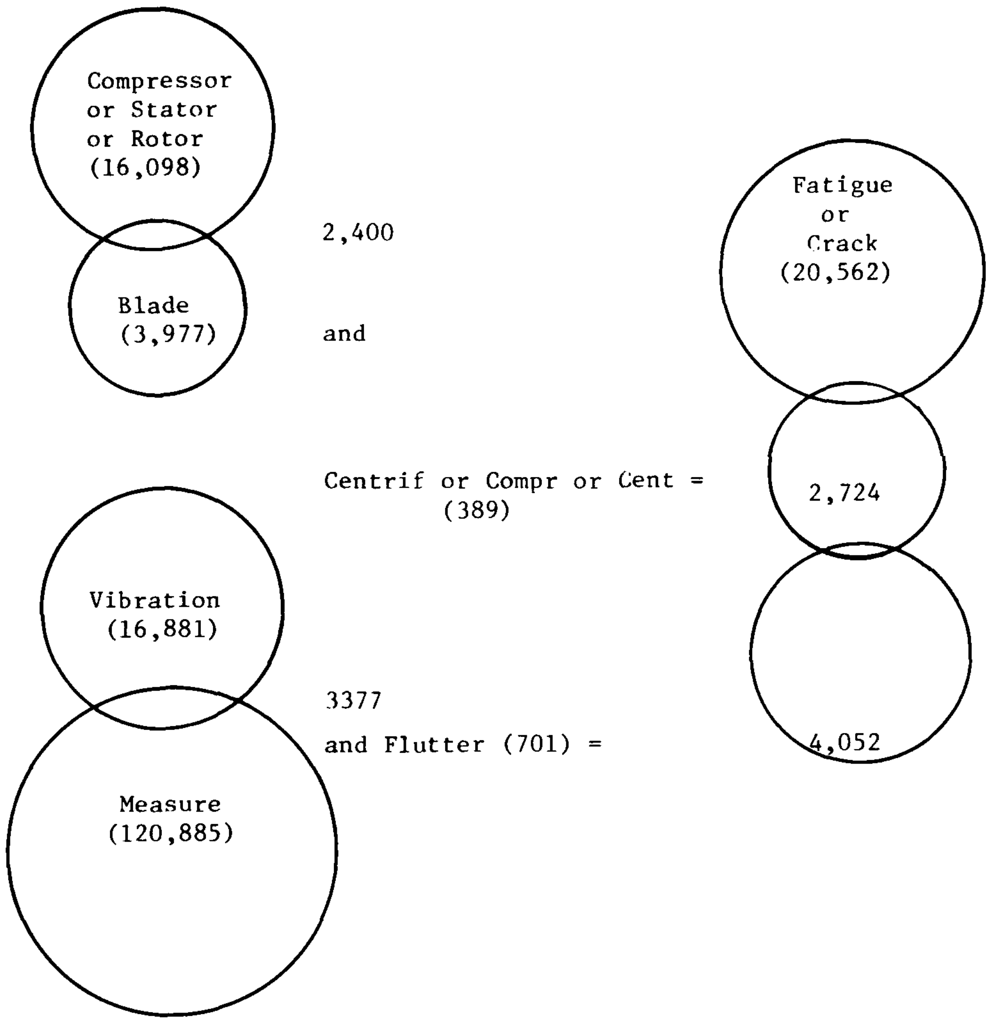
5. Review of Techniques for Monitoring Blades

A literature search was conducted to review work done in the area of compressor blade vibration and cracking. The categories searched were:

- Compressor or Stat or Rotor
- Blade
- Vibration
- Measurement
- Centrif
- Flutter
- Fatigue or Crack

Two literature files were addressed by the search -- NTIS and the Engineering Index. A total of about a quarter of a million articles were checked by the computer in the search.

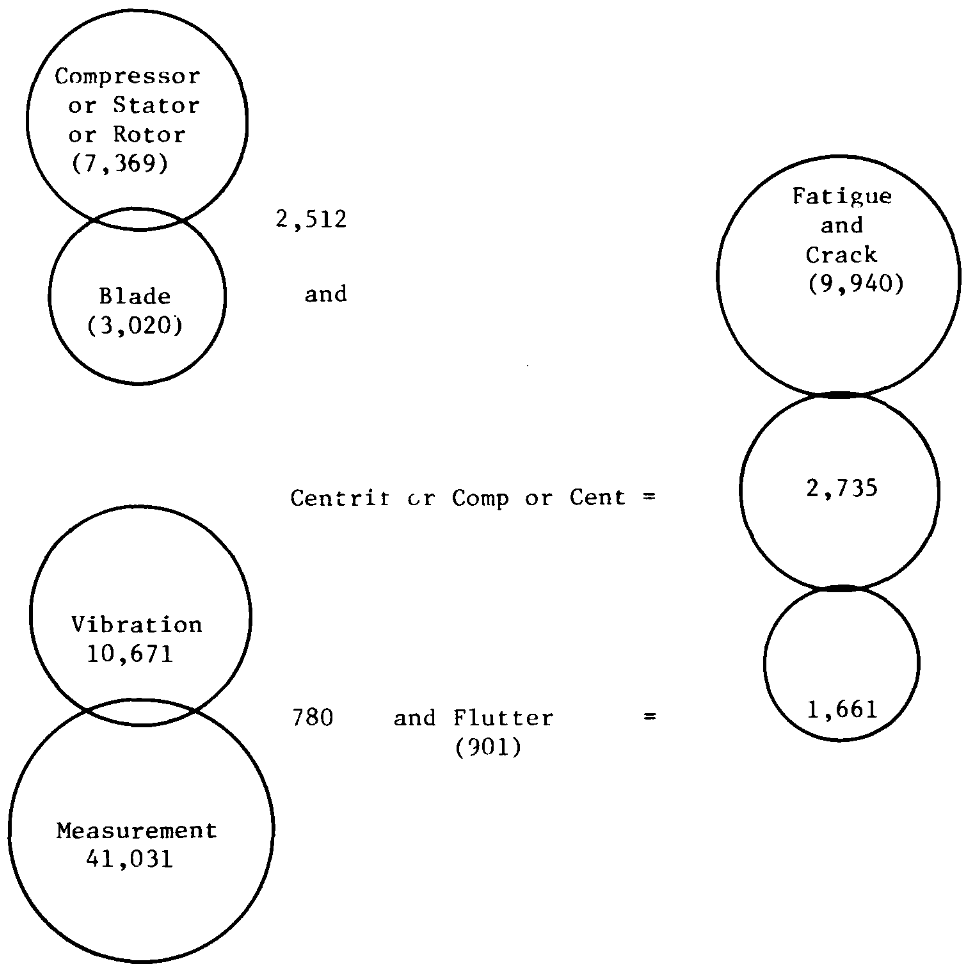
The logic followed in combining these categories is illustrated in Figures 7 and 8. To illustrate the approach, reference is made to Figure 7 Engineering Index Survey. There were 16,098 articles found covering compressor, or stator or rotor and 3977 articles found covering blades. Of these two categories, 2400 articles covered information encompassing both. In addition, 389 articles containing the words centrif or compr or cent were added to the 2400 articles for a total of 2724 articles. Articles containing fatigue or crack and also containing the information sorted in the 2724 articles resulted in 152 articles for which abstracts were obtained. The abstracts were reviewed individually and eight articles finally selected as applicable to the subject of interest. In a similar manner, an additional 32 articles were selected for review, resulting in a total of 40 references for evaluation. While there were not any duplicates in this list of 40 articles, there were some cases where the same technique was described by the same author(s) in two different media. For example, many times an ASME or SAE paper is written that summarizes a full NASA report. The original NASA report was obtained in these cases.



Abstracts Reviewed	Articles Selected	Articles Received
152	8	3
129	18	6

Total Articles Surveyed 179,493

Figure 7 Engineering Index Survey



Abstracts Reviewed	Articles Selected	Articles Received
139	8	1
96	6	5

Total Articles Surveyed 73,287

Figure 8 NTIS Survey

In many cases a final review of the article in the Rensselaer Polytechnic Institute (RPI) Library showed that the article did not cover a dynamic measurement technique and no copy was made of the article.

As a result of the literature search and review of pertinent papers directly, or by reference, twenty-one references were selected as containing information relative to blade monitoring techniques. These are included as references 1 through 21.

a. Technique Evaluation

Almost all of the techniques described in literature were intended to measure blade vibration. One paper by Hegner 16 addresses the problem of damaged blades; however, the application was for aircraft gas turbine engine Foreign Object Damage (FOD). Table 1 lists the more important measurement techniques used or recommended for blade monitoring.

The techniques may be summarized as follows:

1. Measurement of vibration by observing the blade tip. (2, 4, 5, 7, 11, 12, 13, 14, 15, 20, 21)
2. Measurement of blade velocity variation in the plane of the compressor stage. (9, 10)
3. Measurement of blade vibration by scanning the whole blade along the axis of rotation. (3)
4. Measurement of blade damage by scanning the whole blade along the axis of rotation. (17)
5. Detection of blade cracking by the measurement of a change in blade twist. (17)

TABLE 1

SUMMARY OF LITERATURE SURVEY

September 7, 1979

Author	Measurement Used	Experimental Evaluation	Comments	
Kurkov Dicus	Press	Yes	Poorest accuracy	Vibration induced by stalling, amplitude .005 to .06 in. to determine frequency of individual blades
	Strain	Yes	Highest accuracy	
	Optical	Yes	Intermediate accuracy	
Hegner	Multimeter wave inter-ferrometer	No	Leading edge defects	
	Electrometer	No	Not recommended	
	Magnetic field	No	Not recommended	
	Optical	No	Not recommended	
	Remnant mag-netic field	Yes	Detected a struck blade by disap-pearance of mag-netization	
	Eddy current	Yes	Detect damage by twist of blade tip	
Barranger	Eddy current	Yes	Detect disc crack at blade root	
Zablotskiy	Elura device (sensor not described)	Yes	Display of blade vibration using oscilloscope	

TABLE 1

SUMMARY OF LITERATURE SURVEY (continued)

Author	Measurement Used	Experimental Evaluation	Comments
Kulczyk	Laser doppler	Yes	Frequency shift resolution 300m/sec tip speed of $1/0^4$ resolution drops significantly with signal/noise ratio. Beam should reflect off blade face.
Nighten-gale	Microwave Proximeter	No	Detects blade root versus blade tip motion
Roth	Laser	Yes	Amplitude capability of 0.2mm
Staheli	Imbedded mag-net in blade	Rig	Magnetic coil pick-up and demodulation coil pick-up design critical
Bien	Optical inter-ferrometry	Rig	Two reflective spots on blade required. Suited for axial Laser locations.
Nieber-ding Pollack	Photoelectric canning stroboscopic	Full scale engine test	Torsional measurement accuracy of 0.25 degrees

All the techniques of Category 1, except one [7], are indirect measurement of vibration in that they measure the difference in time (or angular position) between when a blade arrives at a given circumferential location vs. the time. The basic measurement concept is shown in Figure 9. The measurement requires a prior knowledge of angular position of the rotor at any instant of time. This can be generated by a gear on the shaft, or to greater resolution an optical encoder or to the highest resolution, a synthesizer producing a large number of pulses for a once-per-revolution input. The second requirement is for a sensor in the compressor case that generates a signal as a blade passes the sensor. If the sensor "sees" the blade when the angular position is exactly θ, then the blade is not deflected at that instant. If the sensor "sees" the blade when the angular position is different than θ ($\theta \pm \alpha$), then the actual deflection of the blade at that instant may be calculated from the value α and the compressor stage geometry. Unless the vibration of the blade is synchronized with rotation speed, the next time the same blade comes to the sensor the deflection will be different. A visual display is used in many of the referenced papers that brightens a spot on the screen of an oscilloscope when the blade passes the sensor. Figure 10 shows the scope screen for the case where there is no vibration and for the case with vibration. Each blade is assigned a vertical position.

Some attempts have been made to measure the actual vibration frequency by measuring the oscillating frequency of the dot on the scope at two carefully controlled shaft speeds [21]. This measurement of frequency is difficult because of the required precision of the measurements. It is also not single valued, however, if the approximate value is known, this is not important. A more complex solution is described in References [2] and [20], in which many sensors are located around the circumference at each stage. Data for each blade is taken at each sensor for every blade. After the test, the data is reassembled into a time trace for each blade.

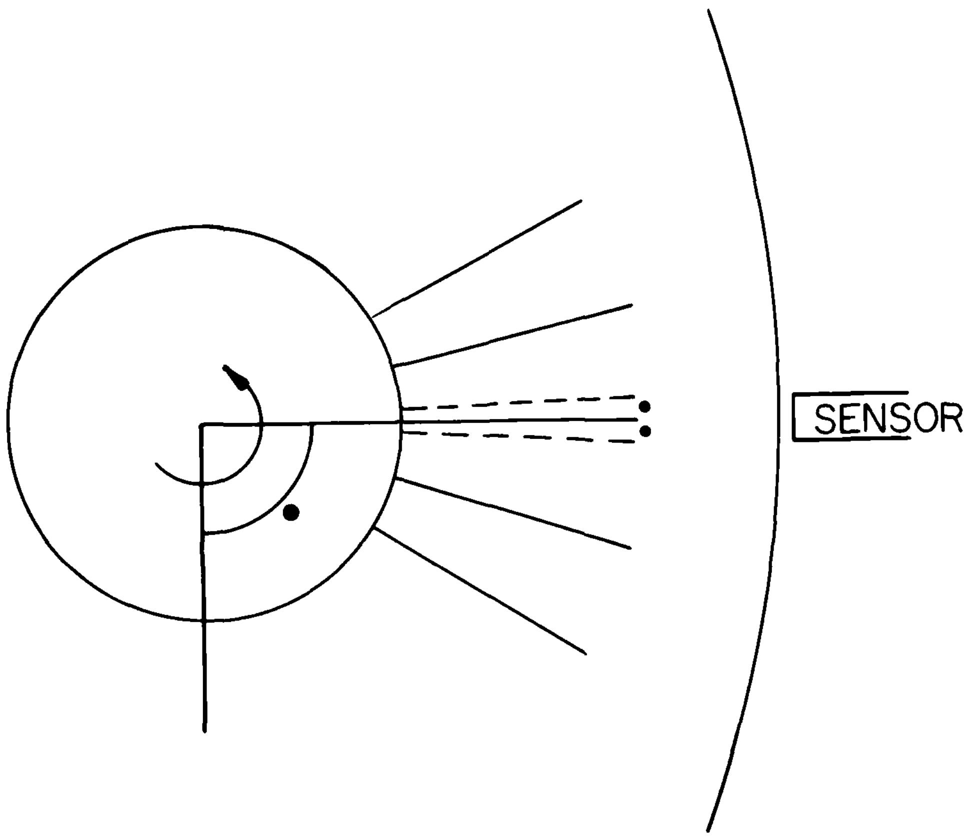
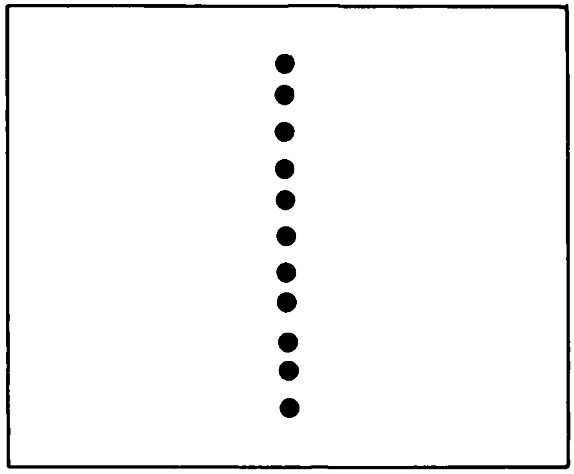
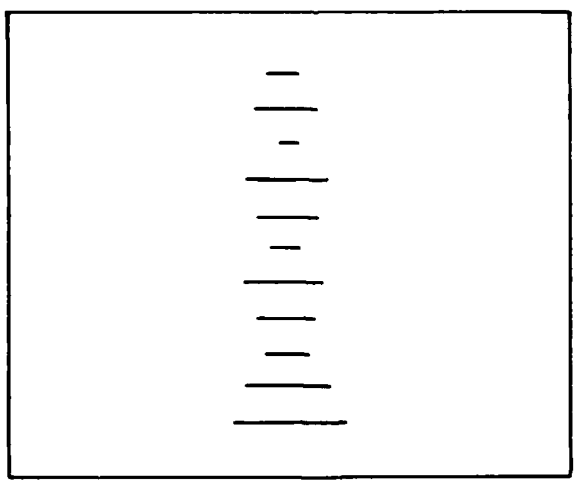


Figure 9 Blade Tip Vibration Detection



NO BLADE DEFLECTION



BLADE DEFLECTION

Figure 10 Blade Tip Deflection Visual Display

Sensors used for picking up the blade arrival pulses were both eddy current and optical sensors with the majority of experimentation using optics. The number of sensors N_s needed is a function of the blade vibration frequency F_n and the rotational frequency F_r and is given by

$$N_s = \frac{F_n}{F_r} \times 2$$

It can be seen that to instrument a multistage compressor with blade vibration up to 1000 Hz would result in a tremendously complicated and costly system.

The one system that measured tip vibration direction [7] required that the blades be made of magnetic material or small magnets be installed on each blade.

References [9] and [11] discuss a technique that directly measures the velocity of the blade by observing the doppler shift in a laser beam scattered by the blade as it interrupts the beam. The vibration velocity is determined for a brief instant of time for each blade by subtracting the measured velocity from the rotational velocity. As such, it is again a sampled system and faces somewhat similar problems to the single port time of arrival concept. Reference [9] contains equations that can be used to calculate the signal-to-noise ratio for the measurement in terms of stage geometry.

Reference [2] describes a system where the whole first stage of a compressor is illuminated by a laser beam via a diverging lens. Retro-reflectors (reflected light always parallel to the incident light) are mounted on certain blades and interference fringes are found between the reflected beam and one split before the diverging lens. While this is a very interesting concept, it is applicable

only to the first stage of a compressor and requires the mounting of retro-reflectors on the blades. It is not applicable to the present problem.

Reference [17] cites among other techniques a millimeter wave interferrometer for detecting first-stage FOD and its limitations to the first stage excludes it from further consideration.

Reference [17] also cites an eddy current measurement system that is designed to detect the twist angle of compressor blades. The theory is that as a crack develops in a blade, the twist angle will change. The initial system tried used two pickups connected in a bridge configuration that was balanced for normal twist. An abnormal twist in a given blade would produce a large signal for that blade. This concept was the only scheme that attempted to measure blade damage via a measurement made in the plane of the compressor stage.

6. Concept Selection

Two possible monitoring concepts emerged out of the study of the papers identified by the literature search. The two concepts were the detection of blade vibration by use of an optical sensor observing the blade tip and the detection of the twist of the blade caused by a blade crack. Neither concept is without problems when applied to the VA 1310 compressor. These will be discussed below.

The last two teardowns of the VA 1310 compressor have included tests in which the natural frequency of each blade was measured. It is expected that if a crack developed in a blade, then its natural frequency would be reduced. Estimates of the magnitude of frequency shifts anticipated prior to failure as discussed in Section 2 indicate .7 to 1.5% change in resonant frequency. Reference [18] shows magnitudes up to 10% depending on crack size and location along the blade length. It would seem, then, that if the vibration frequency can be measured within these

accuracies, then warning of impending catastrophic failure can be given to allow compressor shutdown before serious damage occurs.

The problems cited for the blade frequency measurement included the number of sensors required and the complexity of the data processing system needed to make the measurement. While it is necessary to measure the blade resonant frequency exactly, another alternative was considered rather than propose a large number of sensors be installed per stage.

On the VA 1310, each stage of stators may be adjusted separately. If one sensor is mounted per stage and the oscilloscope presentation is used, then a possible concept would be to adjust the angle of the stator so that the blade could be excited by stator passing frequency. The speed of the rotor would be gradually increased so that the stator passing frequency increased from below the natural frequency to above it. The speed at which the blade becomes excited is then a measure of its resonant frequency. The test could be repeated for each stage. Blade analysis indicated that it was questionable that sufficient energy would be available to excite blades to a reasonable amplitude from stator passing excitations.

Alternately, stalling of a blade could be used as the excitation source. If stalling of a single stage is attempted by adjustment of inlet stators, rotating stall will undoubtedly occur where blades will alternately resonate and not resonate introducing difficulties in the measurement system. It might be possible to stall the whole compressor; however, this could have catastrophic results. Because of this problem, this concept was set aside and the blade twist measurement was examined.

The author, H. R. Hegner, of Reference [17] was contacted to review the measurement of twist. Reference [17] reports that a blade twist of 2° was easily detectable over a length of approximately one inch. This would represent 0.5° for the four-inch probe spacing of the VA 1310 compressor first-stage blade. Mr. Hegner stated that this work was performed using

eddy current probes seven to nine years ago. He felt that sensitivities much greater than 2^{o} could be obtained with the present proximity probe technology. These would include eddy current, optical or very high frequency submillimeter wave radar. References [5] and [20], which measure blade arrival time, are based on very good edge resolution obtainable with optical techniques. The optical system reported in Reference [12] would result in a resolution of 0.079 degrees if applied to the VA 1310 compressor.

The desired blade twist measurement range for the VA 1310 compressor is 0.015^{o} to 0.15^{o}. Since it has been demonstrated [12] that measurement techniques in this range have been made and improved resolution is possible [5, 20], a system based on measurement of blade deflection during operation of the compressor was selected as being superior to present inspection techniques. Requirements for this measurement are discussed in Section 5.

7. Measurement System Concept

The measurement system for monitoring blade twist during operation involves the use of two noncontacting sensors mounted in the compressor case for each compressor blade row (stage). The probes are mounted radially outward at the leading edge and trailing edge of the blade. The probes provide a signal pulse as the blade passes by the probe. In addition to the blade tip monitoring probes, a phase marker probe observing a once-per-revolution pulse from the shaft is required. The time interval between the pulse from the phase marker probe and the leading and trailing edge probe pulses from each blade is indicative of the blade position for each blade. Differences between arrival time of the leading and trailing edge probe pulses indicate the degree of blade twist. Differences in arrival time between phase marker pulses and both blade probe pulses indicate blade bending. The time resolution of the system depends upon the magnitude of blade tip deflection which, in turn, is dependent upon the crack size to be monitored.

The remainder of the monitoring system entails collection of the time delay pulses for each blade in a stage row as a function of speed. The data collected in a microprocessor is then transferred to a computer for analysis and comparison with previously stored data. The requirements for this portion of the data acquisition system are also a function of the measurement time resolution of the tip measurement probes which dictate the time of data collection and available time for data transfer. The overall system concept is illustrated in block diagram in Figure 11.

a. Probe Requirements

In order to define the time resolution for the monitoring system, the relevant characteristics of the VA 1310 compressor are tabulated in Table 2. The data is tabulated for the largest (first stage) blade and smallest (tenth-stage) blade to encompass the range of characteristics that will be encountered in the system.

Obviously, it would be desirable to provide the capability of detecting a crack as it begins to initiate. The associated blade deflections with such a small crack would not be discernible. System resolution, therefore, depends upon the resolution capability of the probe. Since the probe is located a distance away from the blade tip, it cannot discern a single point of the target but rather an area of the target. If the target area is considered a circle of diameter D (circle of resolution) and it is desirable to detect a blade deflection of 0.5×10^{-3} in., it might be anticipated that the target area (D) would have to be 0.5×10^{-3}in. The circle of resolution of the probe, however, is not that stringent. The probe pulse is fed to a level detector which is activated at a preselected threshold level. The leading edge of a typical pulse signal is illustrated in Figure 12. If the threshold level is set midway up the pulse height with a tolerance of plus-or-minus 10 percent, then the circle of resolution can be expanded an order of magnitude or greater depending upon the probe design. For a

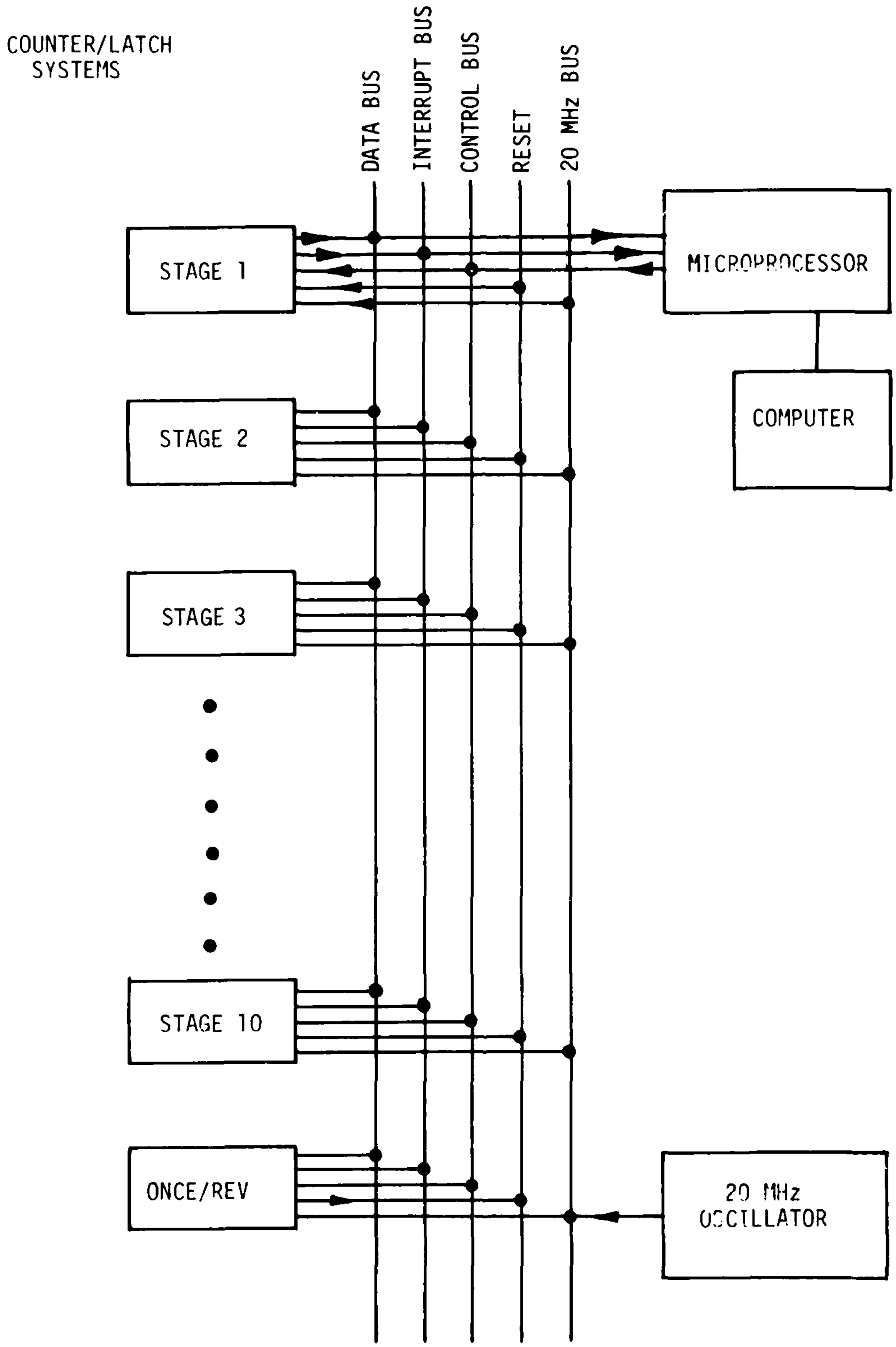


Figure 11 Bus Structure

TABLE 2

COMPRESSOR BLADE CHARACTERISTICS

	First Stage Low Pressure	Tenth Stage High Pressure
Rotor hub diameter (in.)	38	38
Maximum blade length (in.)	7.305	3.293
Maximum number of blades/stage	37	47
Speed (RPM)	3,600	3,600
Angular velocity (rad./sec.)	376.99	376.99
Angular velocity (deg./sec.)	21,600	21,600
Blade tip speed (in./sec.)	9,916	8,404
Angle between blades (rad.)	0.17	0.1337
Angle between blades (deg.)	9.73	7.66
Time between blades (μsec.)	450	355
Minimum time between blades (μsec.)	413	334
Minimum crack size (in.)	0.0071	0.0295
Angular twist associated with minimum crack size (rad.)	.00025	.000308
Angular twist associated with minimum crack size (deg.)	.014	.018
Tip deflection with minimum crack (in.)	.0005	.0005
Time resolution required (nanosecs.)	50	50

NOTE: Numbers given refer to the low pressure stages with high pressure stage numbers in parentheses.

- Blade thickness may be ignored if the electronics operate on the leading edge of the pulse.

- Blade mounting tolerance is plus or minus 0.14° (0.18°). If two successive blades are at either end of the tolerance band, then the angular spacing of the blades would be reduced by 0.28° (0.36°)

- As a minimum blade failure will occur as crack size exceeds .145 in. (0.100) with a resultant blade angle twist of 5 (14) times the perceptable twist angle. Depending on direction of crack propagation, crack size could increase beyond .44 inch before failure occurs.

- If a crack of 0.442 occurred, blade twist angle would increase to 0.29° (5.65°) reducing the spacing between blades to 7.23° (8.92°). The time between blade arrival would then be 334 (413) μsec.

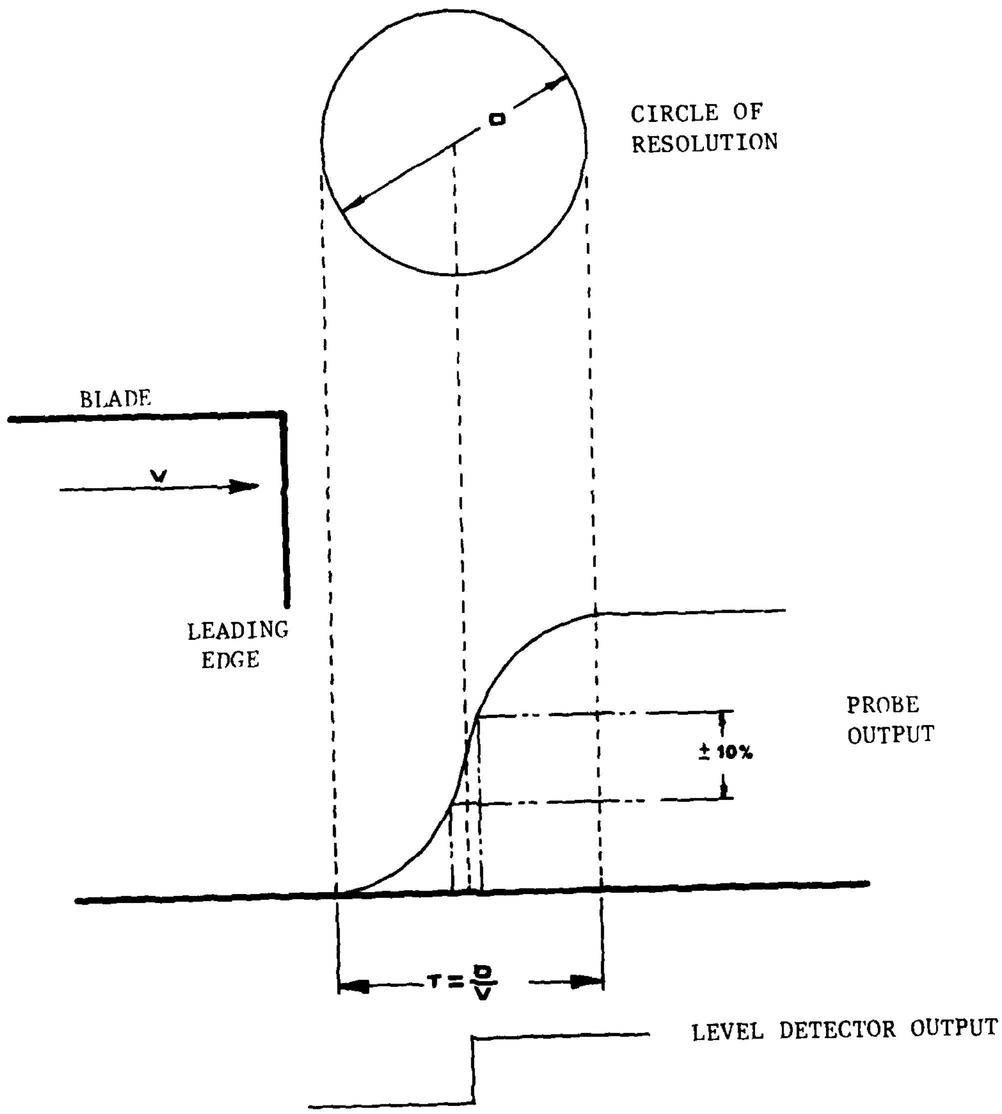


Figure 12 Probe Response

50 nanosecond time resolution, a 5 nanosecond rise time of the level detector is required. Since there are tradeoffs between the probe resolution, repeatability of the level detector and the time resolution, it is best to specify for the probe system only the time resolution required of 50 nanoseconds. That allows the actual system designer to select the tradeoffs best suited to him.

5. Counter/Latch System

Figure 13 illustrates the probe concept for a single stage. The system is comprised of two probes per stage (A and B) and a third probe (C) to generate a once-per-revolution pulse from the shaft. A 20 MHz crystal oscillator is shown that serves as a 50 nanosecond clock to provide the required resolution. The output of the probe level detectors is connected to the counterlatch system. In this system the time count between the once-per-revolution phase probe pulse and the blade tip probe pulse is held for transfer to the microprocessor by means of a latch and register. Data transfer is accomplished between arrival pulses of successive blades. The time delay count for every blade is latched and transferred during one revolution.

Figure 14 illustrates the counterlatch system in greater detail. The two latches, A and B, are transparent latches. This means that every time a latch pulse is received, the count on the 16-line input is held. There is no clear pulse needed to reset the latch. Once the count is latched, the data ready flag is raised. This tells the central microprocessor that data is available at this latch. When the microprocessor is ready to accept data, it sends a byte select word to the multiplexer. This byte select code also enables the tri-start driver at the MUX output which puts data on the data bus.

A special case of the counter/latch module is the once-per-revolution counter shown as Latch C in Figure 13. It is basically the same as

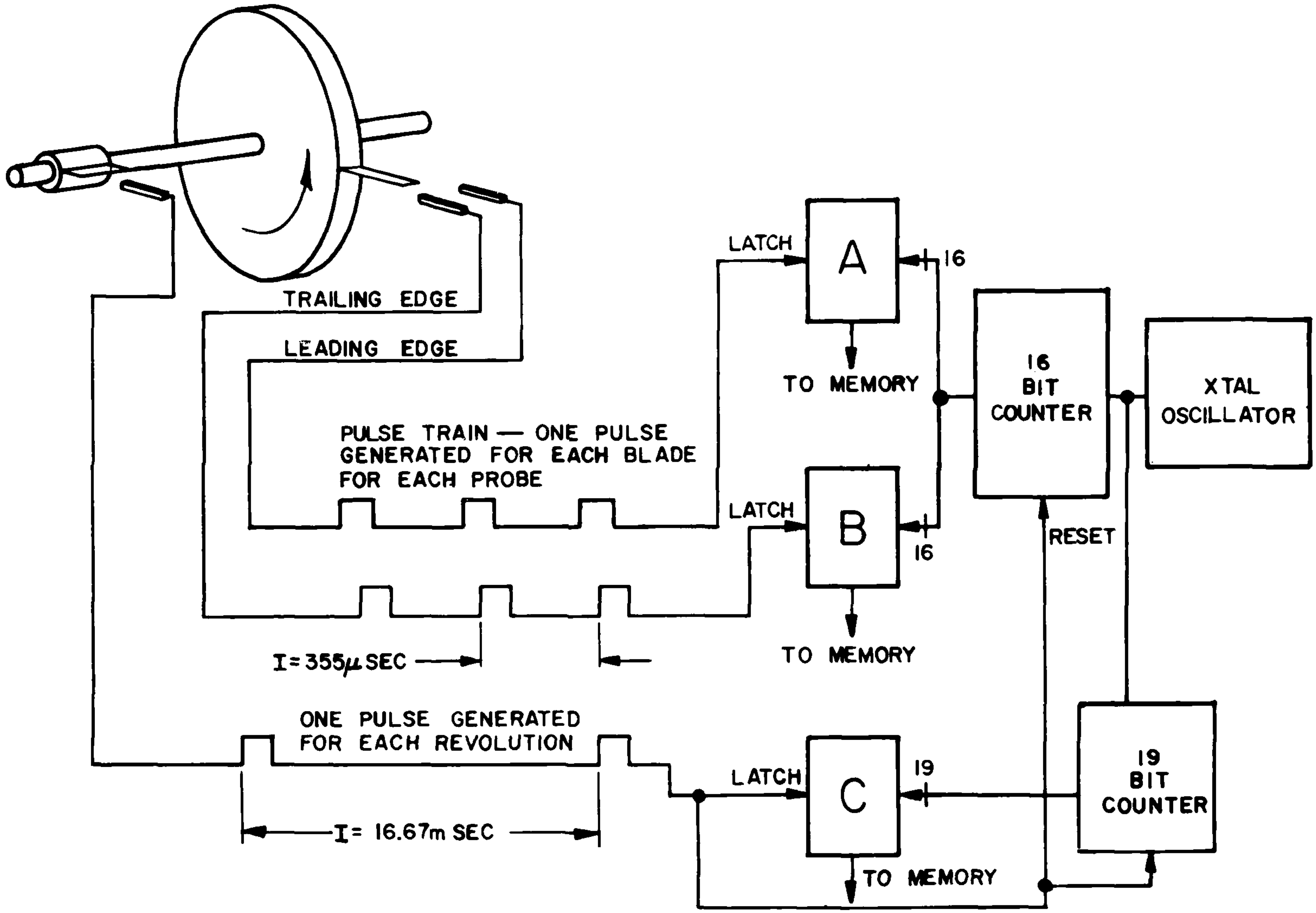


Figure 13 System Concept

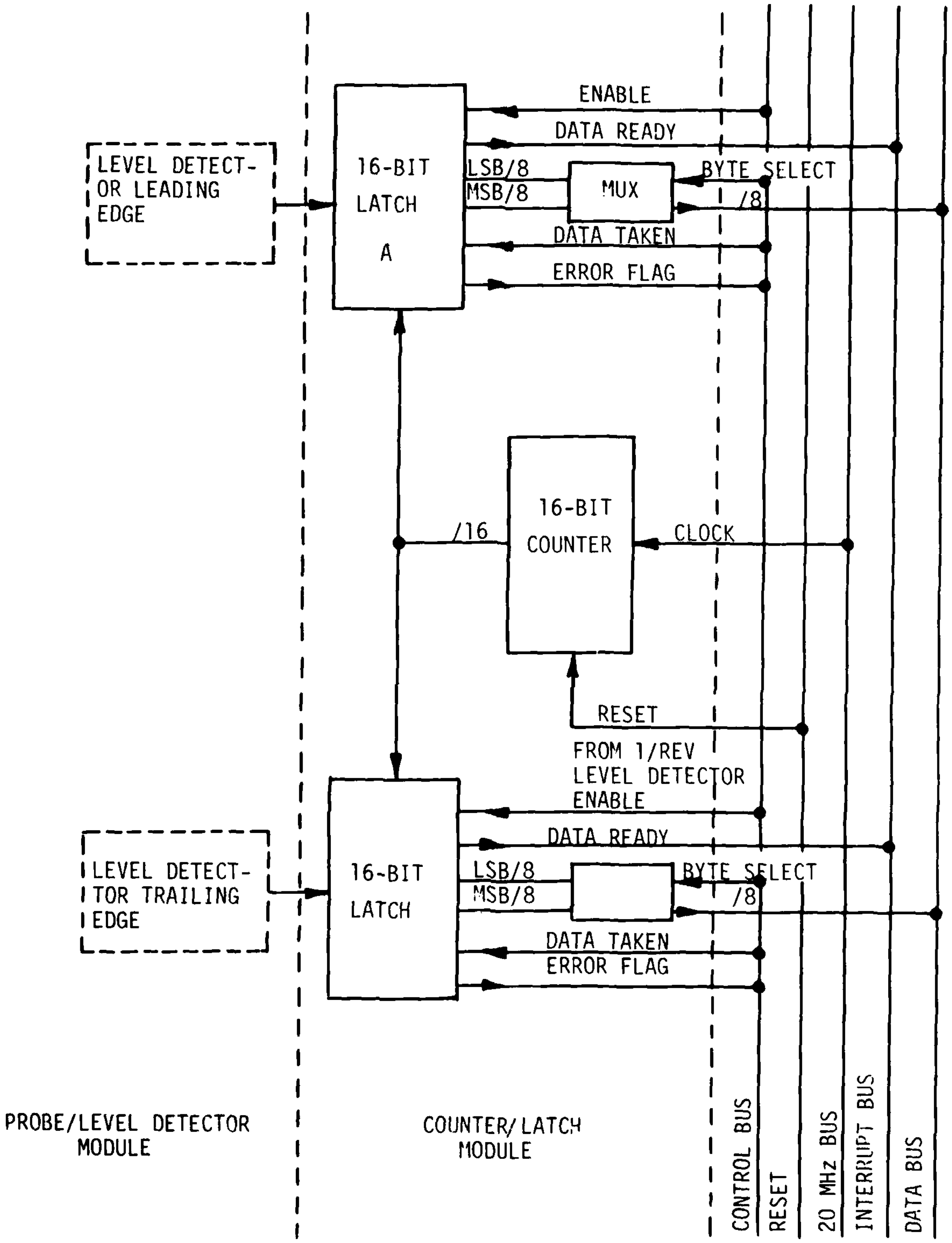


Figure 14 Counter/Latch System

the system shown in Figure 14, except that the counter and latch have 19-bit resolutions. This means that the data must be transferred out in three bytes rather than two so that the MUX must accept a two-bit code. While this data transfer only takes place once-per-revolution, or every 16.6 milliseconds, the data transfer must still take place between two blade arrival pulses. In fact, during the interval between two blade arrival pulses in which the once-per-revolution pulse is received, seven bytes of data must be transferred to the microprocessor; two from A latch, two from B latch, and three from C latch.

The multiplexer (MUX) is required since the microprocessor is an 8-bit microprocessor and therefore must take the 16-bit count in two 8-bit bytes. Once the first byte has been received by the microprocessor, it then selects the second multiplexer code and takes the second byte. After the second byte is received, it raises the data taken flag to the latch from which it has taken data. This flag is *not* required to unlatch the latch; its only function is to insure that the system is operating. For example, if a new latch pulse is received before the data received flag is raised, then the error flag is raised. At the completion of a test, the test is considered valid only if neither the A or B error flag is raised.

c. Microprocessor System

The microprocessor system is illustrated schematically in Figure 15. The microprocessor as envisioned is an 8-bit processor and may be any that will perform the required function within the time requirements. Three I/O (input/Output) ports are shown. One for the data bus, one for the control bus, and one to communicate with the host computer. A special input port is the interrupt port that is tied to the three data ready lines. The clock simply establishes the speed of the microprocessor (up to its limit). The program to operate the system is contained in the PROM. Data is stored in the RAM.

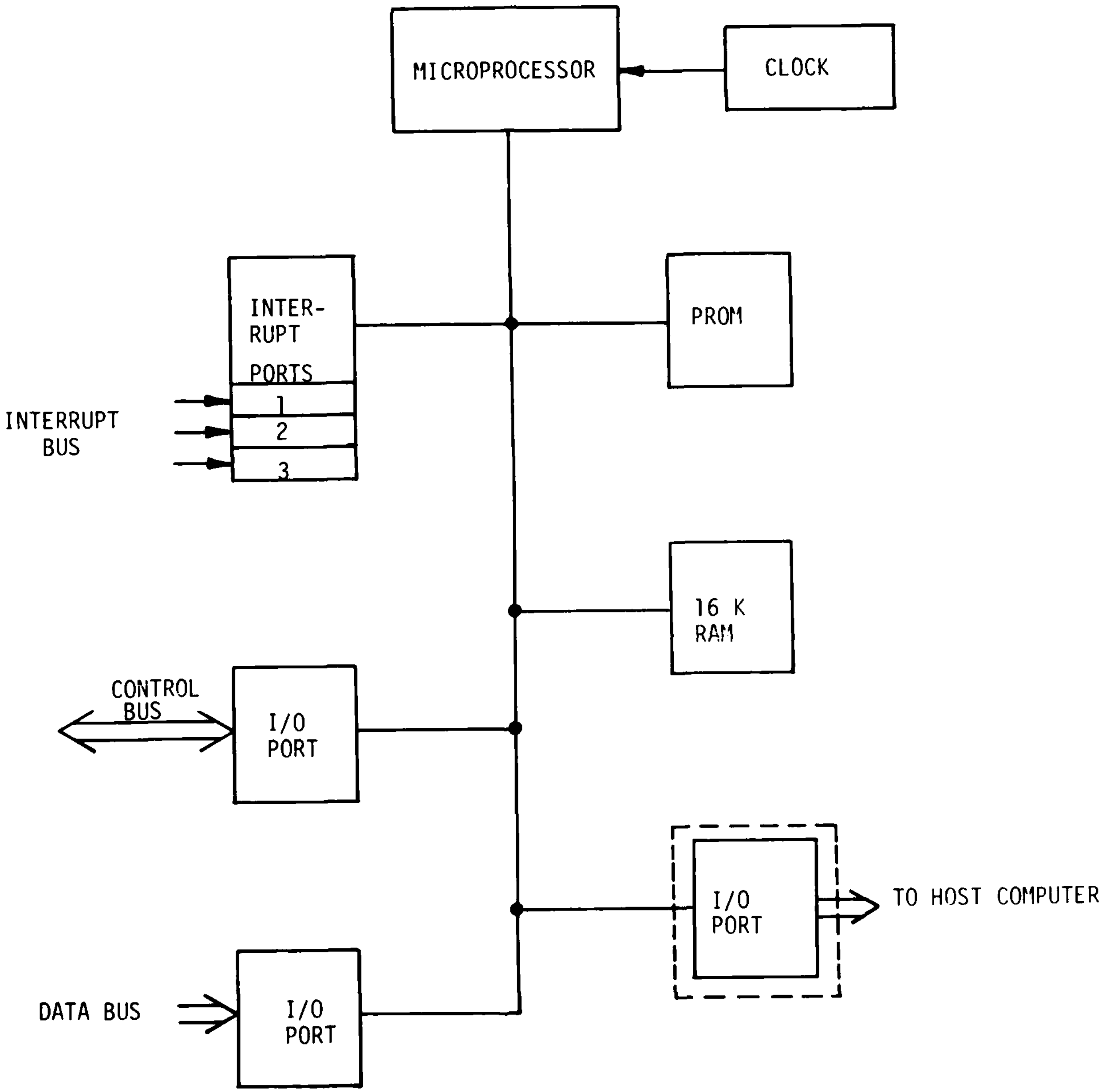


Figure 15 Microprocessor System

The amount of data storage can be incrased up to almost 64 k bytes (64 k bytes -- the PROM memory length).

The I/O port communicating with the host computer can either be bit serial (RS232) or byte serial (IEEE Bus). This will depend upon the physical length of transmission and the amount of data to be transmitted. The host computer can be used to control the complete measurement sequence. If this is not desired, then an additional I/O port would be used that would allow this control to be accomplished via a set of switches or a keyboard.

The operation of data gathering would be as follows: manual control would input the number of revolutions to be averaged; next, the stage to be checked would be input. At this point, the microprocessor would send enable signals to the proper board. Next the start experiment would be selected. The microprocessor would then monitor the C interrupt line, which indicates that a once-per-revolution pulse has occurred. Following this occurrence, all three interrupt ports are given equal priority. Suppose the next occurrence is a blade pulse latching the A counter. An interrupt is obtained at port one. On receiving this interrupt, the microprocessor jumps to a routine to take data from this latch. The least significant byte is selected and the tri-start driver is enabled. When the byte is received, it is stored in a given memory location and the next byte is transferred and stored. The routine is then excited. If another interrupt occurred, or when it does occur, it is then received in almost the same manner except the memory location is different. The next time an interrupt occurs on the C line (one/revolution), a routine is called that gets three bytes of information from its 19-bit latch. This routine does one other thing. It checks to make sure that N values have been stored for the A and B latch when N is the number for the selected stage. If this is true, the revolution counter is incremented which increments the memory location, and data from the next revolution is ob-

tained. If N does not equal the number of blades, then the revolution counter is not incremented and the new data overwrites the last set.

When data from the preselected number of revolutions have been received, the process stops. Data may now be sent to the host computer. A number of tradeoffs are available in transferring data from the microprocessor regarding the amount of preconditioning being done in the microprocessor. It is possible for the microprocessor to average the data of each blade for every revolution and transfer the average data in lieu of all data. For blade twist data, the microprocessor could transmit only the differential time count between leading and trailing edge probes to the host computer. The advantage of performing one or more functions in the microprocessor is to reduce data transfer times. Transfer of data to the host computer could be by RS232 or IEEE standard formats. If a blade rate of 9600 were used, transmission would occur at the rate of 600 bytes per second. Depending upon the approach selected, transmission times could reach 20 seconds. Selection of the most suitable system is therefore a system design consideration.

d. Overall System Design

The overall configuration was shown in Figure 11. The system consists of three buses, plus one line carrying the 20 MHz clock and one carrying the once-per-revolution reset pulse. The buses are:

DATA Bus
Control Bus
Interrupt Bus

For the ten-stage compressor, there are actually eleven counter/latch systems. The eleventh being the special 19-bit latch for the counts-per-revolution. The 20 MHz oscillator is connected to each of these

counter/latch systems. The 1/rev signal is used to reset all counters when the 1/rev pulse is received. As such, it originates in the 1/rev (actually the associated probe) system and is sent to all other systems. The interrupt bus consists of three lines. Line one connects the data ready lines from all B latches to the microprocessor. Line two connects the data ready lines from all B latches to the microprocessor. Line three connects the data ready lines from the C latch on the once-per-revolution board to the microprocessor. Depending on which stage is enabled, the data ready signals from that stage will cause an interrupt in the microprocessor. Note, the one-per-revolution is always enabled. The control bus consists of eight lines (8-bits). Four are used to determine the stage address. The remaining four lines are used to formulate up to sixteen different control commands. The data bus contains eight lines for the 8 bits of information.

As discussed in Section b, the output of each latch is a MUX that would allow data to be taken from the latch one byte (8 bits) at a time. The output of the MUX will be a tri-start driver. The three states of this drive are the two binary states (1 & 0) and one state in which the output impedance is very high. The high output impedance decouples the drive from the data bus. When the driver is enabled, the output impedance goes very low and the device now is capable of driving the data bus. This allows many drivers to be connected to the data bus with the central microprocessor controlling which driver it wants data from.

3.0 SUMMARY

Current techniques used to verify the integrity of compresor blading in the Allis Chalmers VA 1310 compressor, entail disassembly of the machine and verification of blade natural frequencies by measurement. The procedure is followed by magnetic particle inspection of the blades for cracks not detected by measurement of the blade natural frequencies. Both inspection techniques assume that if a crack develops in the blading, it will not progress to failure between teardown and inspection intervals. Under certain events such as blade instability or ingression of a foreign object, the validity of this assumption is questionable. Furthermore, the inspection technique imposes considerable time and expense to conduct a teardown and rebuilding of the compressor. An inspection procedure that does not require disassembly, or a continuous blade monitoring system, would offer obvious advantages over the present blade inspection techniques.

In order to determine the feasibility of developing a blade inspection technique that would eliminate compressor disassembly, a preliminary analysis of blade deflections was conducted to determine the sensitivity of blade tip motions to blade damage. In addition, a literature survey was conducted to establish the state-of-the-art of blade monitoring techniques.

The results of blade analysis resulted in the following conclusions:

a. Blade root cracking will induce blade tip deflections. The largest deflections occur by angular twisting of the blades.

b. Shifting in blade resonant frequency relative to crack size is not as sensitive as blade tip deflection relative to crack size.

The literature survey produced the following results:

a. Measurement of blade vibration through the use of blade tip monitoring sensors has been demonstrated by a number of investigations.

b. Blade vibration measurements were primarily directed toward determination of blade resonant frequencies.

c. Blade twist measurements have been demonstrated as a means of detecting blade damage. These measurements were also accomplished by non-contacting tip deflection monitoring probes.

As a result of these studies, it was concluded that a blade deflections monitoring system would be most suitable for detecting impending blade failure in the VA 1310 compressor. Based on these findings, a system was described that utilizes non-contacting blade tip monitoring probes, a microprocessor and a host computer. The requirements of all elements of the system are state-of-the-art components. Prior to constructing the system described in this report, further definition is required in the following areas:

- Laboratory evaluations should be conducted on the probe type and resolution, to establish the minimum blade deflections that can be measured.
- A study should be conducted on the method of probe installation in the compressor.
- A host computer should be selected and interface requirements between microprocessor and computer developed.

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